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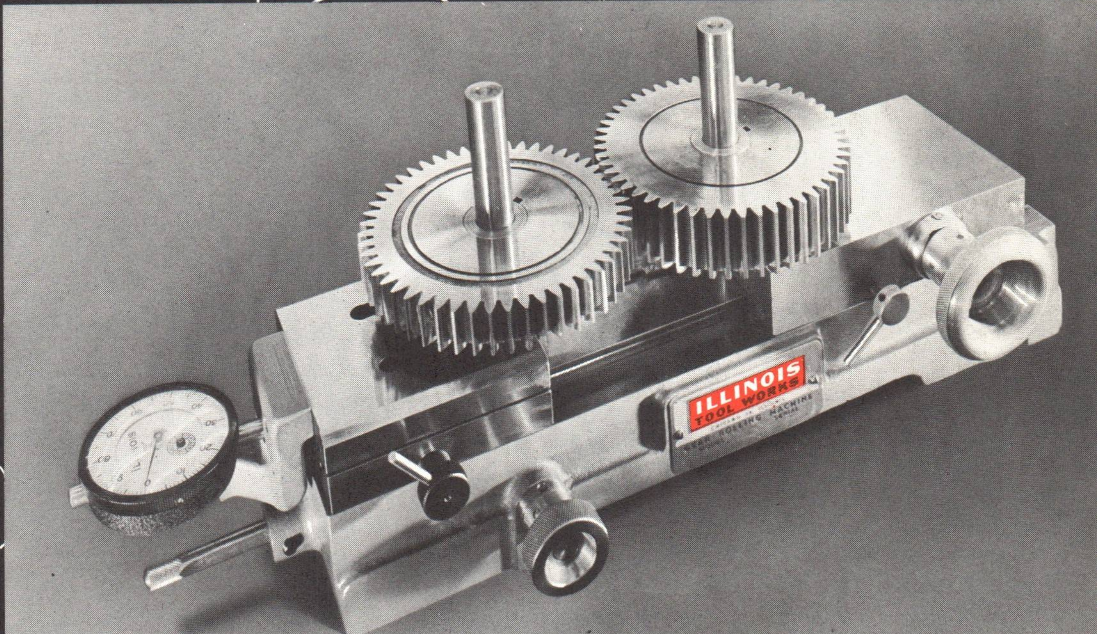
TOOL WORKS

Know Your MASTER GEARS



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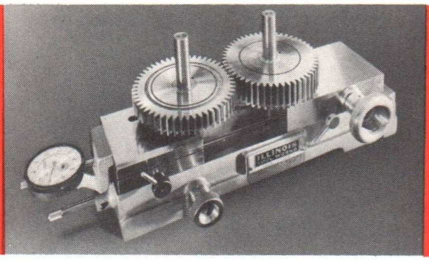

Know Your MASTER GEARS

Master gears play an all-important role as the basic gaging element in modern functional gear checking, the practice of testing gears by the way they roll or function. Historically, master gears were the first method of gear checking; today, in the most up-to-date gear departments, master gears and rolling fixtures are still the policemen of the process.

In functional gear checking, the gear to be checked is rotated with the master gear in intimate metal-to-metal contact. If involute profile, tooth spacing, helical lead and concentricity are perfect in both master and gear, the two will roll together with no variations in center distance. If errors are present in any one of their elements, variations in center distance will occur. Size diameter and runout will be reflected at their true value. Involute profile, tooth spacing and helical lead errors will also be indicated at a distorted but magnified ratio, depending upon the pressure angle. Errors in the master gear itself, regardless of their nature, confuse the accuracy picture of the gear and subtract from the manufacturing tolerances.

Inaccuracies in the master gear are very costly in production. In all probability, the rolling check has been assigned a tolerance for indicator variation. Errors in the master contribute to this indicator movement and thus leave a smaller tolerance for the production processes. They keep operators from getting identical results on successive checks. In the way these master gear errors increase or cancel those of the production gears, they may cause a good gear to be rejected or a bad one to be accepted. *Great accuracy is therefore the most important characteristic of master gears.*

In addition to being highly accurate, these are the other important characteristics of master gears:

They should check production gears over the entire active portion of the involute. (*See Fig. 1.*)

Their exact tooth thickness should be known for center distance checking.

Their design in general should help to produce and retain the highest degree of accuracy.

The term "master" may be misleading to many people because it implies that the master gear is a duplicate of a production gear. Some parts, cams for instance, whose proportions are difficult to express in simple, measurable dimensions, are frequently checked and compared against a master cam of identical size. A master gear, however, need not be like the gear it represents. As a matter of fact, good master gears seldom are true duplicates of their representative production gears.

The design of a master gear has to fulfill the basic requirements of checking a certain production gear over its entire width and to the full active depth of its involute. In the past, and to quite an extent today, this design problem was simply solved by making the master gear exactly like the mating gear but slightly wider on its face.

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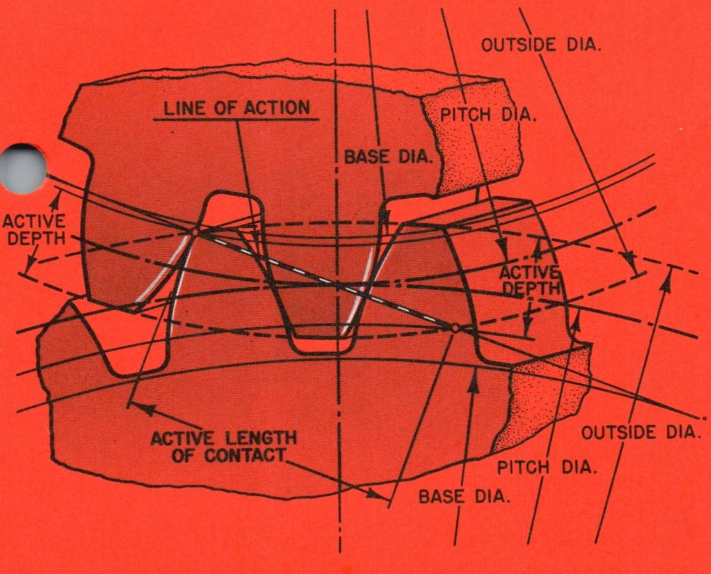


FIG. 1

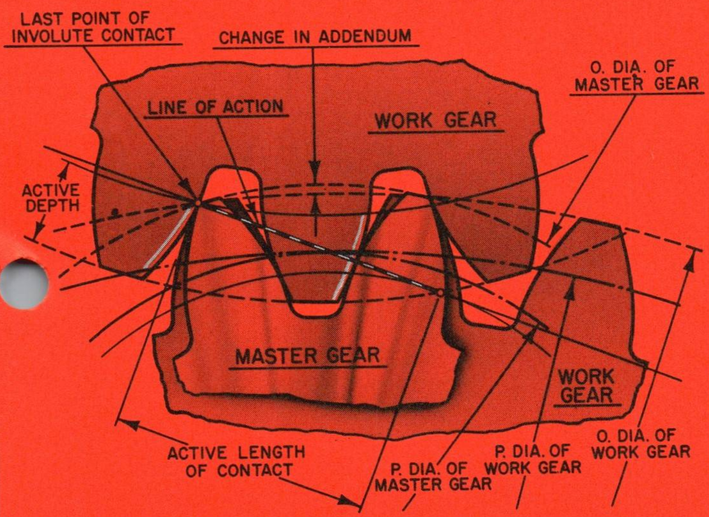


FIG. 2

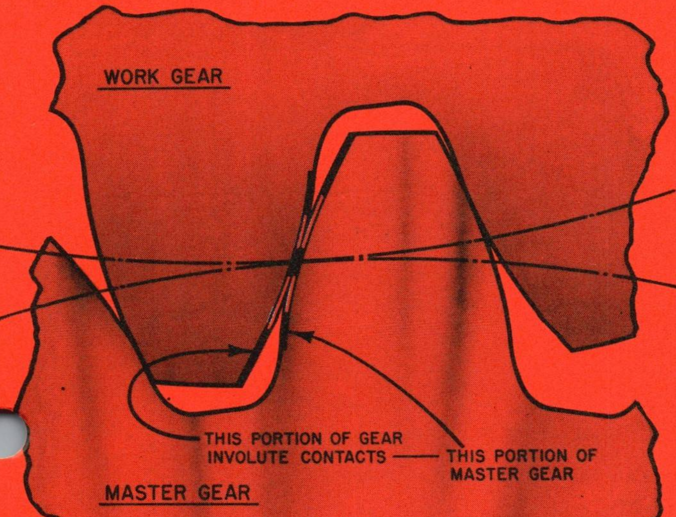

WORK GEAR
THIS PORTION OF GEAR INVOLUTE CONTACTS
THIS PORTION OF MASTER GEAR
MASTER GEAR

FIG. 3

This practice is entirely justified whenever the general proportions of the mating gear are also ideal for the master gear. It should definitely not be followed when it leads to diameters and number of teeth which are not in the interest of maximum accuracy at minimum cost.

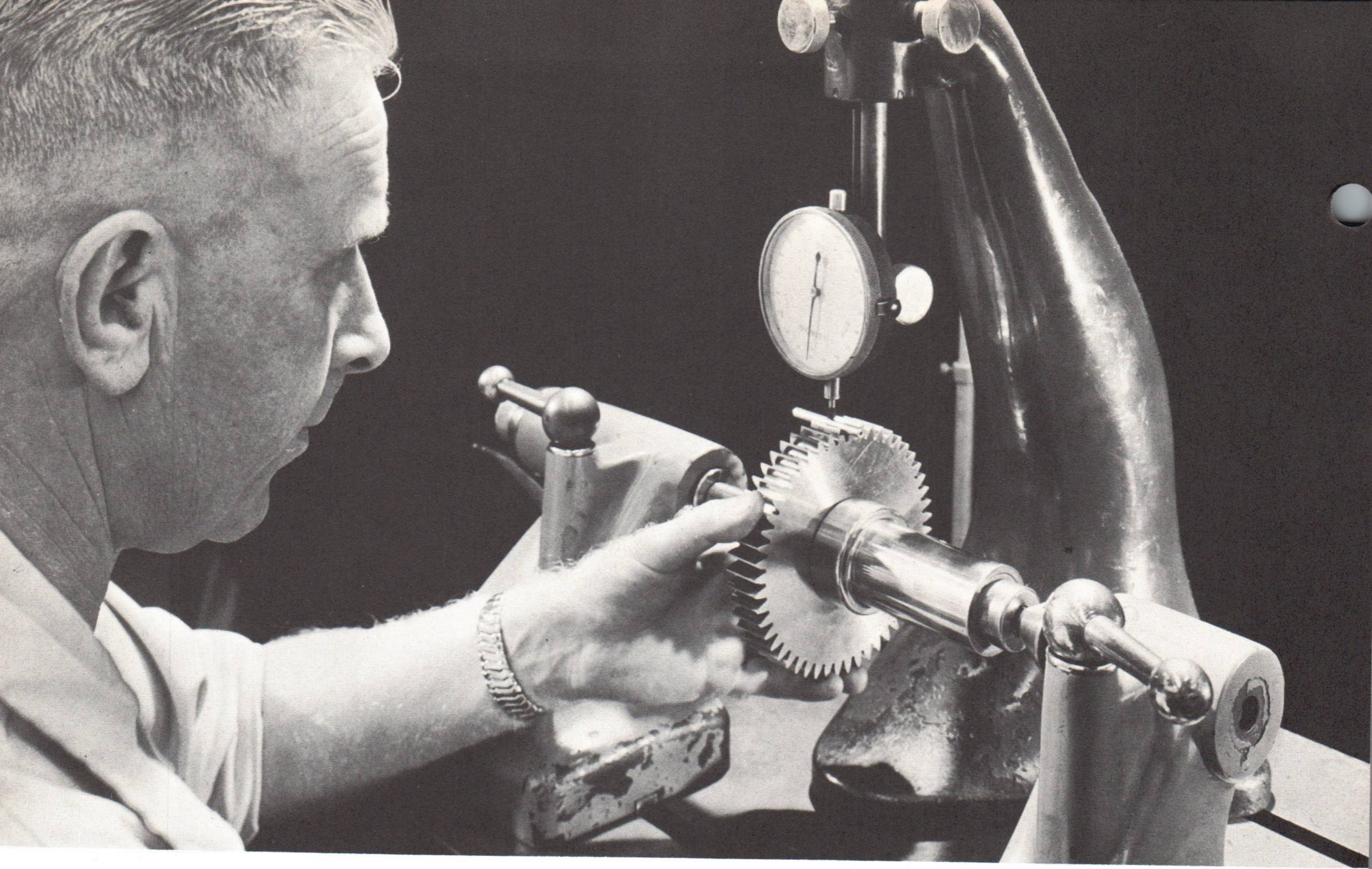
The calculation of the active involute profile and the last point of contact with the mating gear is very simple for those who are familiar with gear geometry. It is equally easy to determine the addendum of a master gear which may not be like that of the gear and yet contacts over the same area. (See Fig. 2.) Those not familiar with these calculations should leave the design of master gears to the manufacturers of these gears. They should realize, however, that complete data on both mating gears is required for the proper design of a master.

There is one other design consideration which is strictly in the interest of the *continued* accuracy of the master gear. The portion of the involute near the base diameter is most difficult to hold accurate and most subject to wear. Therefore, the ideal master gear should be designed, if possible, to avoid any contact below approximately 5° PA. (See Fig. 3.)

In addition to correct gear design requirements, the relative proportions of diameter, face width and hole size are just as important on master gears as they would be on any other part which is to be highly accurate and is to retain that accuracy over a long time. Master gears depend for location upon their holes and faces and these should be proportioned to insure continued accuracy.

The foregoing rules might imply that all master gears should be designed for one specific gear only. Table 1-A, however, shows suggested standard sizes for 20° PA gears. Master gears made to these specifications meet all criticism for minimum size. At the same time they are large enough not to be contacted below 5° PA by most gears.

Theoretically, of course, the OD or addendum of a master gear can only be designed for one



STANDARD COARSE PITCH MASTER GEARS

DIMENSIONS

20° P.A.

DP	NO. OF TEETH	PD	FACE WIDTH	HOLE
4	20	5	1¼	1¼
5	20	4	1¼	1¼
6	24	4	1¼	1¼
7	21	3	1	1¼
8	24	3	1	1¼
9	27	3	1	1¼
10	30	3	¾	1¼
12	36	3	¾	1¼
14	42	3	¾	1¼
16	48	3	¾	1¼
18	54	3	¾	1¼

TABLE 1-A

14½° P.A.

DP	NO. OF TEETH	PD	FACE WIDTH	HOLE
4	32	8	1¼	1¾
5	35	7	1¼	1¾
6	36	6	1¼	1¼
7	35	5	1	1¼
8	40	5	1	1¼
9	36	4	1	1¼
10	40	4	¾	1¼
12	36	3	¾	1¼
14	42	3	¾	1¼
16	48	3	¾	1¼
18	54	3	¾	1¼

TABLE 1-B

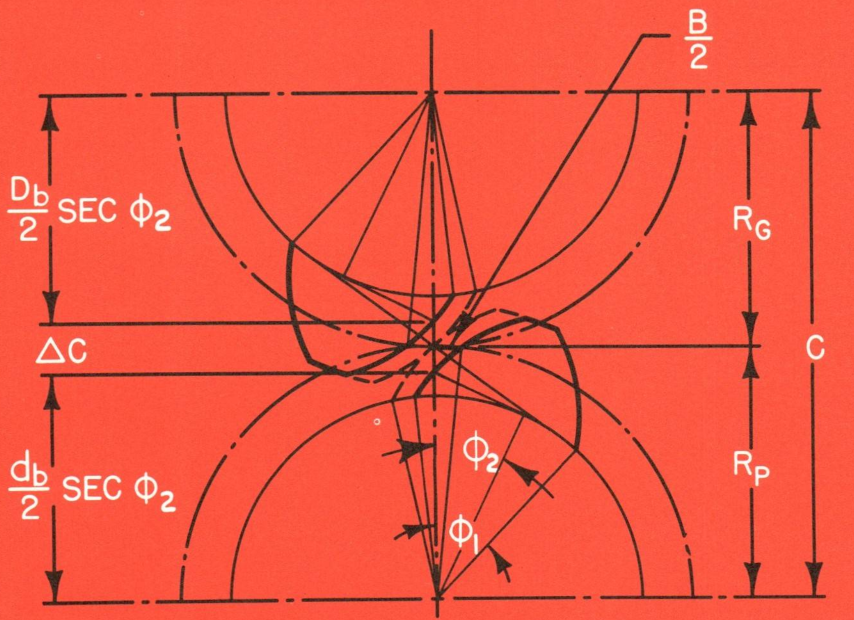
*Helical Master Gears will conform approximately to the above values but may have variations dependent upon the helix angle. Use the nearest pitch diameter to obtain the number of teeth.

specific gear. Other gears of the same basic data, however, may be checked by the same master gear. In such cases contact will extend slightly beyond or short of the active depth. The change is very gradual but may, in some cases, necessitate grinding the OD of the master gear.

If the PA of the gears is smaller than 20°, the number of teeth in the master gear should be increased. Table 1-B indicates the ideal size for master gears of 14½° PA.

With the exception of some instrument gears, most gears are made with backlash. Therefore, in metal-to-metal contact with a gear of standard tooth thickness they will roll at reduced center distance. In other words; most production gears are undersize gears. In those instances where the

$$\text{INV } \phi_2 = \text{INV } \phi_1 - \frac{B}{2C} \qquad \Delta C = C(1 - \text{COS } \phi_1 \text{ SEC } \phi_2) \qquad C_R = C \pm \Delta C$$



B = BACKLASH
D_b = BASE DIAMETER OF GEAR
d_b = BASE DIAMETER OF PINION
ΔC = CHANGE IN CENTER DISTANCE
C = THEO. CENTER DISTANCE
C_R = ROLLING CENTER DISTANCE
R_G = PITCH RADIUS OF GEAR
R_P = PITCH RADIUS OF PINION
ϕ_1 = PRESSURE ANGLE
ϕ_2 = ROLLING PRESSURE ANGLE

FIGURE 4

functional or rolling check is also a check of the size of the gear, a calculation of the reduction in center distance must be made.

It is fairly common practice to approximate this correction by applying the following formula:

$$\text{Center Distance Correction} = \frac{\text{Total Backlash}}{2 \text{ X } \tan \text{PA}}$$

Total backlash is to be taken as the difference between the circular pitch and the sum of the arc tooth thickness of both master gear and gear.

Actually, undersize or oversize gears establish rolling diameters other than the nominal pitch diameters. The exact size of these rolling circles not only depends on the amount of undersize but also varies with the number of teeth of the gear. The correct way of establishing center distance is shown in Figure 4. The amount of backlash varies not only with the pitch but also with the specific use of the gears, and no standard figures for the reduction in center distance can be established. Under average conditions the reduction in tooth thickness for say a 6 DP 14½° PA gear is approximately .003. If the gear had 28 teeth and were to be checked by a master gear of 24 teeth and of exactly standard arc tooth thickness, $\left(\frac{1.5708}{\text{DP}}\right)$ the center distance correction by the exact formula would be .00585 while the correction by the approximate formula would be .0058. Such a small deviation is immaterial under most circumstances.

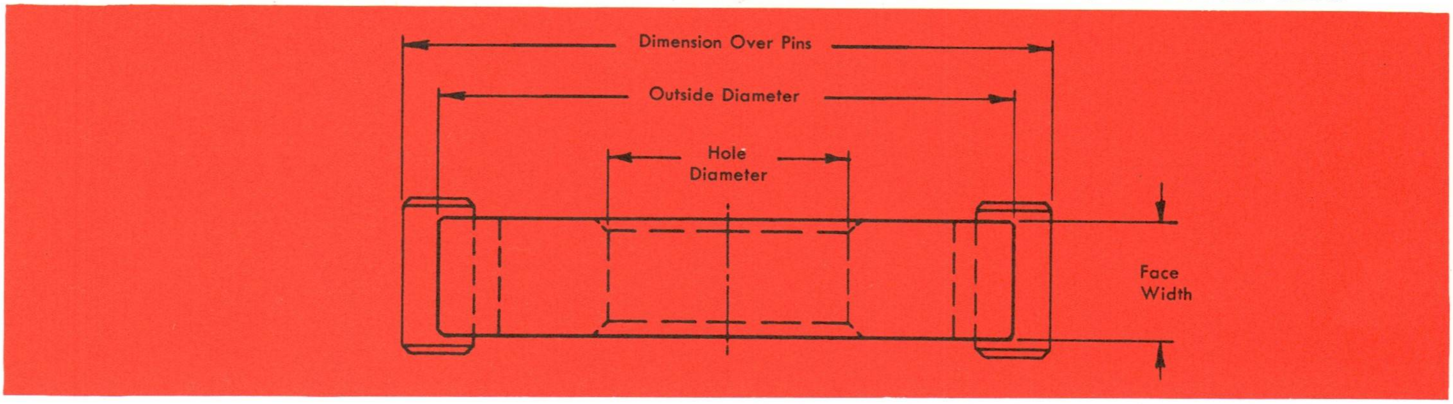
It is important to realize that any master gear having the same number of teeth as the mating gear, but with standard tooth thickness and addendum, will roll at reduced center distance and contact the gear beyond the last point of contact with its real mate unless its addendum is corrected.

There are two classes of master gears—Class A (Precision Size) and Class B (Commercial Size). The terms "precision" and "commercial" apply only to the tolerances on size. All other tolerances are the same for either class.

The reason for the existence of a separate Precision-Size class of master gears is that a few gear manufacturers want the tooth thickness of their masters to be closely held. They may either want their master gears to be interchangeable without having to make a correction in the center distance setting of their rolling fixtures or they may want to dispense with the calculations which are required when checking for size with an over, or undersize master gear. In either case, specifying a Class A (Precision Size) gear is a convenience rather than a necessity.

A close tolerance on size works a considerable hardship on the manufacturers of master gears because it becomes necessary to have all other tolerances under control by the time size is reached. If any one element should be outside its tolerance, it is impossible to regrind without going undersize. Therefore, Class A

GROUND COARSE PITCH MASTER GEAR TOLERANCES*



Tolerances listed are in agreement with those established by MCTI.

CLASS "A" (PRECISION SIZE)

DIAMETRAL PITCH		4-5.999	6-11.999	12-19.999
Active Involute Profile (TIR)		.0003	.0002	.0002
Dimension Over Pins	13° to 19° PA	±.0014	±.0011	±.0007
	Over 19° PA	±.0010	±.0008	±.0005
PITCH DIAMETER		4.250 or less	4.251-6.250	6.251-8.250
Runout (TIR) **	13° to 19° PA	.0007	.0008	.0009
	Over 19° PA	.0005	.0006	.0007
SPACING	Max. variation between adjacent teeth	.0002	.0002	.0003
Lead or Parallelism (TIR)		.0001 per inch of face with minimum of .0001 for central 80% of face width with additional .0001 tolerance allowed adjacent to each face.		
Face Runout (TIR)		.0001 per inch of diameter		
Hole Diameter		+.0002 −.0000		
Outside Diameter		±.0010		
Arc Tooth Thickness: (Transverse)		Taken at the diameter obtained by the number of teeth divided by the transverse diametral pitch.		

CLASS "B" (COMMERCIAL SIZE)

DIAMETRAL PITCH		4-5.999	6-11.999	12-19.999
Active Involute Profile (TIR)		.0003	.0002	.0002
Dimension Over Pins		±.0050	±.0050	±.0050
PITCH DIAMETER		4.250 or less	4.251-6.250	6.251-8.250
Runout (TIR) **	13° to 19° PA	.0007	.0008	.0009
	Over 19° PA	.0005	.0006	.0007
SPACING	Max. variation between adjacent teeth	.0002	.0002	.0003
Lead or Parallelism (TIR)		.0001 per inch of face with minimum of .0001 for central 80% of face width with additional .0001 tolerance allowed adjacent to each face.		
Face Runout (TIR)		.0001 per inch of diameter		
Hole Diameter		+.0002 −.0000		
Outside Diameter		Compensated within ±.0010 to final dimension over pins.		
Arc Tooth Thickness: (Transverse)		Taken at the diameter obtained by the number of teeth divided by the transverse diametral pitch and is compensated to the final dimension over pins.		

TABLE 2

*For tolerance values on helical master gears use the normal Diametral Pitch and normal Pressure Angle

**Measured at approximate Pitch Diameter

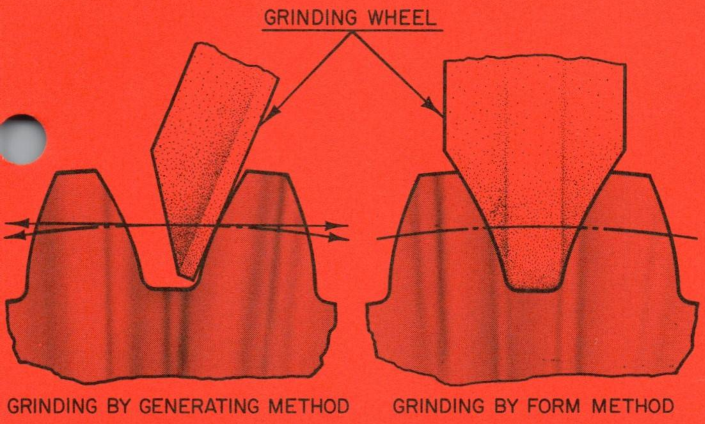
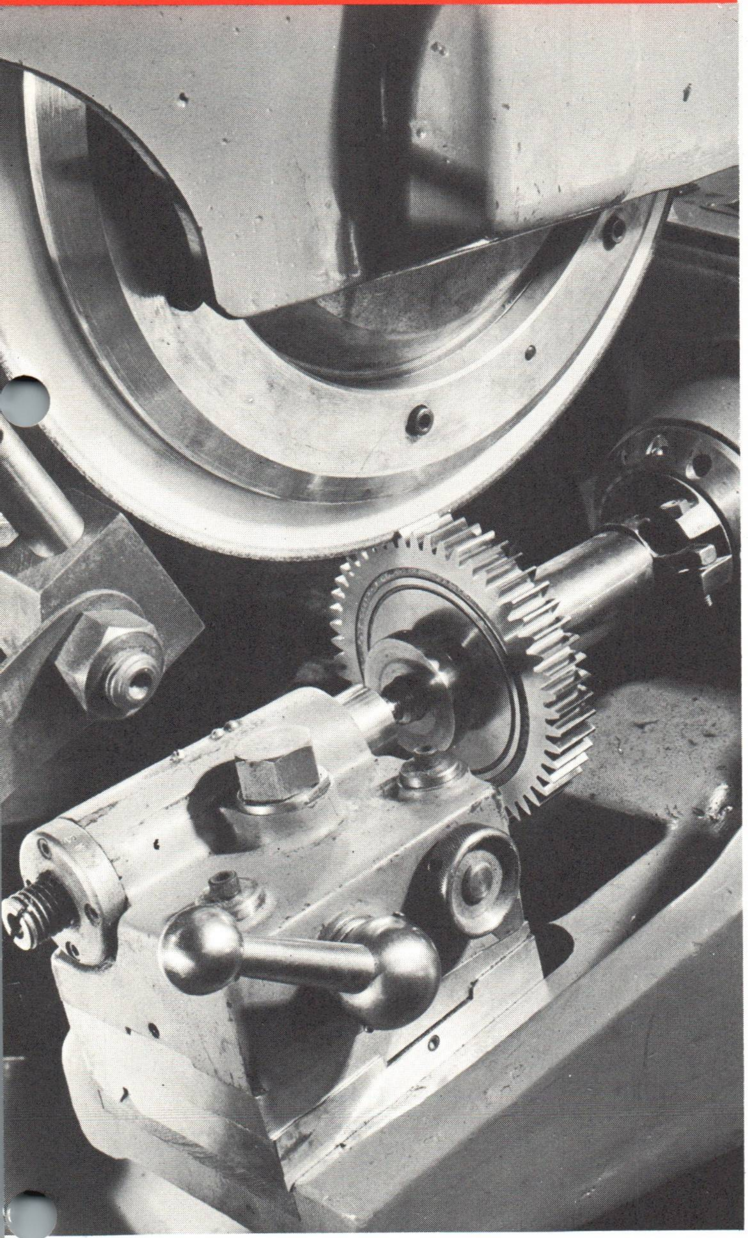


FIG. 5



master gears are more expensive.

Basically there are two methods of grinding master gears—by generating and by form grinding (See Figure 5). The generating method produces correct involutes regardless of tooth thickness. In the case of form grinding, however, the shape is dressed on the grinding wheel and is only correct for one specific position of the grinding wheel. All Illinois Tool Works master gears are ground by the generating method.

Table 2 shows the tolerance charts used by Illinois Tool Works for Classes A and B master gears, which are those adopted as standard by the Metal Cutting Tool Institute. Master gears may be better, but no worse than shown in Table 2. Regardless of class, the exact arc tooth thickness is marked on each master gear. This tooth thickness figure is taken into the calculation of the center distance correction in a very simple way.

The amount of correction depends upon the actual arc tooth thickness of both the gear and the master. As previously mentioned, most gears have less than standard tooth thickness to provide backlash. The arc tooth thickness of the master is etched upon the master gear.

Through the use of a graph like that shown in Figure 6, this correction becomes one of mere subtraction or addition from the nominal center distance, $\left(\frac{\text{PD Gear} + \text{PD Master}}{2}\right)$.

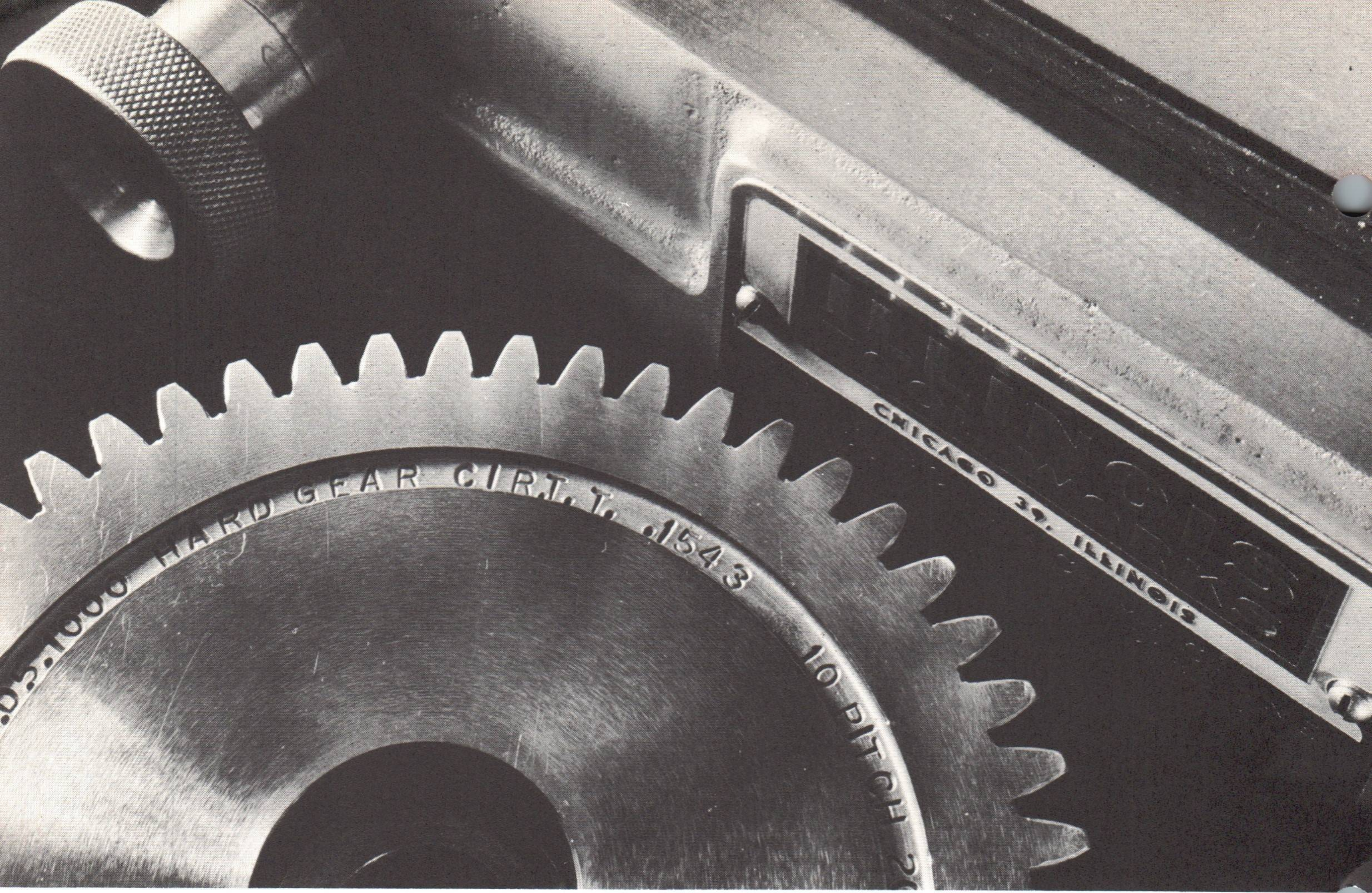
In Figure 6, one scale indicates the difference of the sum of both arc tooth thicknesses from the circular pitch, that is:

$$\text{Circ. Pitch} - (\underset{\text{(gear)}}{\text{arc TT}} + \underset{\text{(master)}}{\text{arc TT}})$$

The other scale indicates the center distance correction. For example:

A work gear with

80 teeth
10 DP
8″ PD
14½° PA
Actual arc tooth thickness .1552



rolling with a master of the following characteristics:

40 teeth
10 DP
4″ PD
14½° PA
Etched arc tooth thickness .1585

$$\text{Circular Pitch} = \frac{\pi}{\text{DP}} = \frac{\pi}{10} = .3142$$

$$\begin{aligned}\text{Tooth Thickness Correction} &= .3142 - (.1552 + .1585)\\ &= .3142 - .3137\\ &= .0005\end{aligned}$$

For 14½° PA, Fig. 6 indicates a center distance correction of .001 (see Fig. 6).

It is entirely possible, particularly in the coarser pitches where the backlash may be great, that the difference between the circular pitch and the sum of the arc tooth thicknesses will exceed the values in Fig. 6. In such instances it is permissable to divide the difference by the nominal center distance, spot the value on the vertical scale of Fig. 6, follow it out to the PA line, read the center distance correction and multiply that correction again by the nominal center distance. For example:

A work gear with
32 teeth
4 DP
8″ PD
20° PA
Actual arc TT—.3827

rolling with a master of the following data:

20 teeth
4 DP
5″ PD
20° PA
Etched arc TT=.3939

$$\text{Circular Pitch} = \frac{\pi}{4} = .7854$$

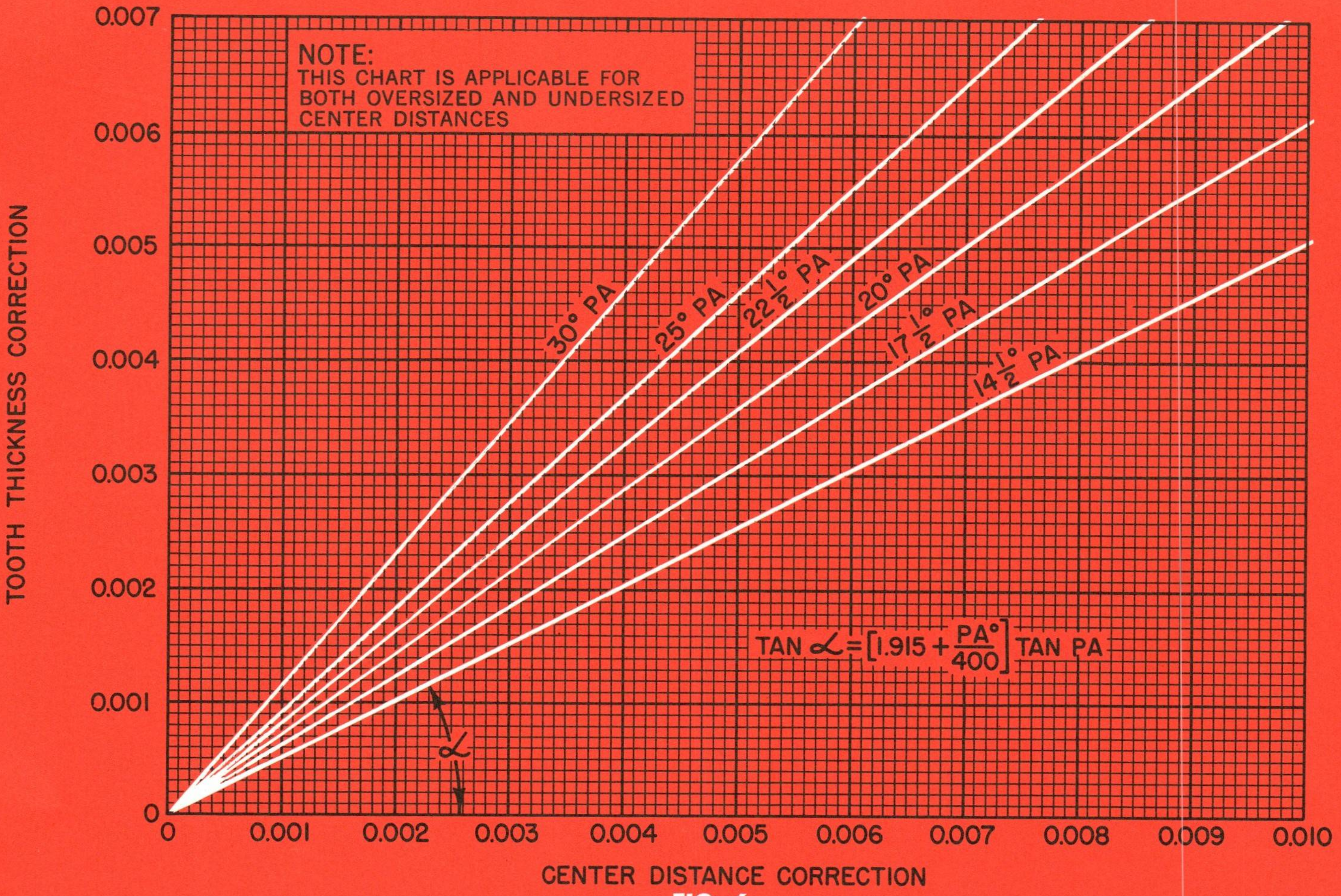


FIG. 6

Sum of arc tooth thickness of gear and master $= .3827 + .3939 = .7766$

Difference $= .7854 - .7766 = .0088$

Nominal center distance =

$$\frac{\text{PD (gear)} + \text{PD (master)}}{2} = \frac{8 + 5}{2} = 6.500$$

$$\frac{.0088}{6.500} = .00135$$

Center distance correction = .0019
(from Fig. 6)

Total center distance correction $= .0019 \times 6.5 = .01235$

Naturally, Fig. 6 may also be used for the reverse process of determining the tooth thickness of the work gear from the measured center distance. The formula is

Arc TT (gear) = Circ. Pitch — (Arc TT (master) + Dif. From Fig. 6)

For example:

If in the previous example the actual center distance between master and work gear had been found to be 6.487, then the center distance correction would be

Nominal center distance — Actual center distance

or

$$6.500 - 6.4877 = .0123$$

Since .0123 is outside the range of Fig. 6, divide by the nominal center distance

$$\frac{.0123}{6.500} = .0019$$

For a value of .0019 on the horizontal scale and 20° PA Fig. 6 gives a total tooth thickness difference from circular pitch of — .00135

Actual difference is $.00135 \times 6.500 = .0088$

Arc TT = circ. pitch — (arc TT + Actual dif.)
(gear) (master) (see above)

Arc TT = $.7854 - (.3939 + .0088)$
(gear)

Arc TT = $.7854 - .4027$
(gear)

Arc TT=.3827

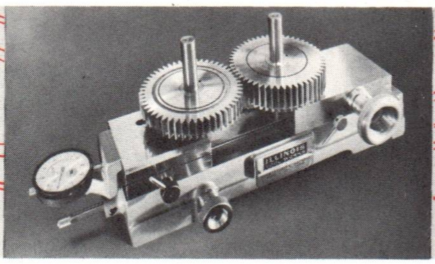
Fig. 6 shows only the lines of the most popular pressure angles. Other PA lines may be entered by making their slope,

$$\text{Tan}\,\alpha = (1.915 + \frac{\text{PA}^\circ}{400})\ \text{Tan PA}$$

Such additional pressure angle lines will be particularly useful for helical gears because the transverse PA must be used in calculating center distance corrections or tooth thicknesses. The arc tooth thickness which is marked on a helical master gear is in the transverse plane.

Good master gears are subjected to a series of treatments designed to minimize any subsequent distortions. It should be remembered, however, that all gears when rolling together, are subject to some wear. Therefore, master gears should be inspected from time to time. Regrinding is rarely advisable because the cost is almost equal to a new gear. Besides, it is usually the reference surfaces, that is, hole and faces, which are most subject to wear.

Master gears are expensive gaging elements. Their original cost, however, is small compared to the expense they may save if correctly designed, accurately made and properly used.



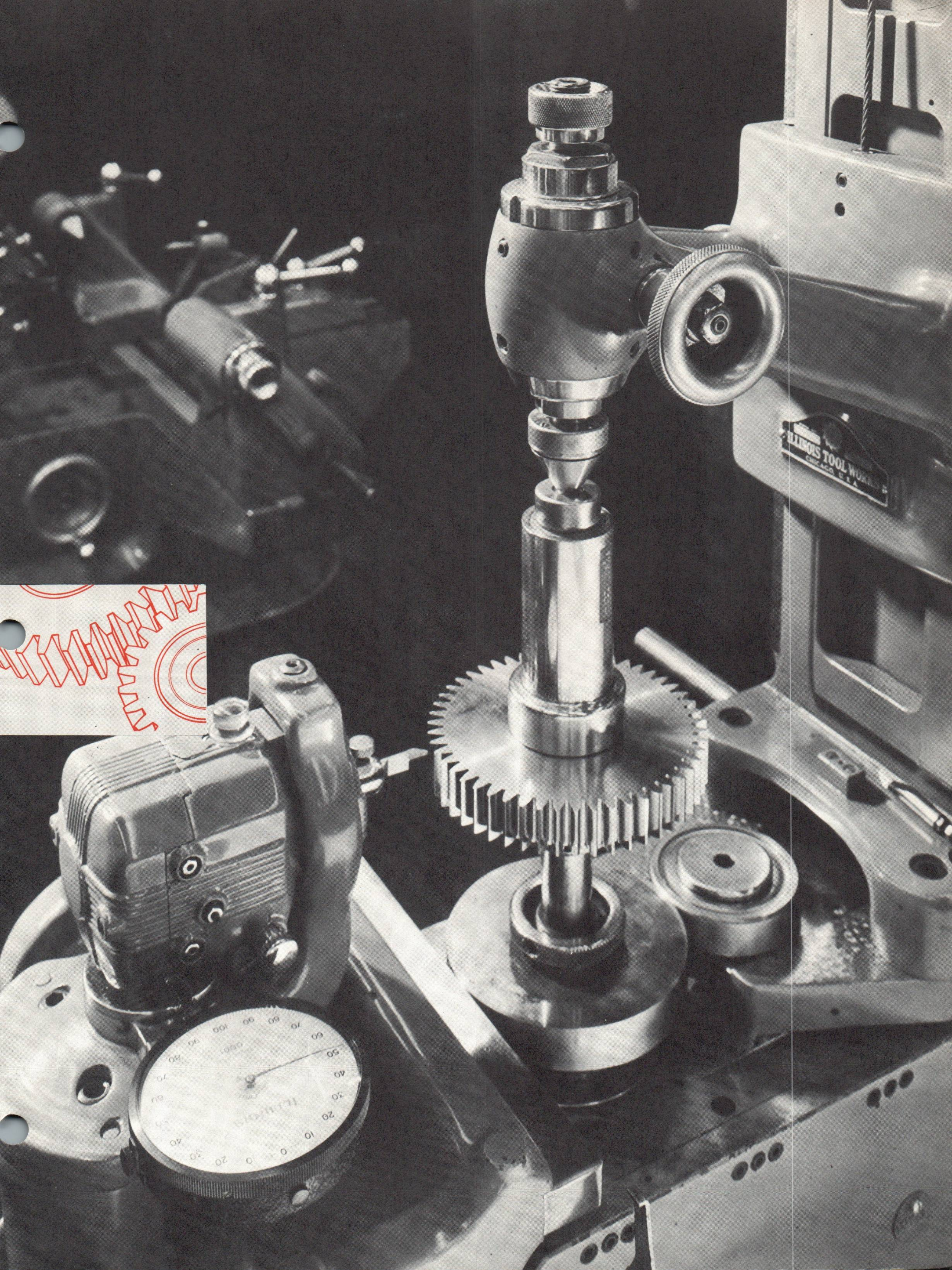
SUMMARY

The design of master gears, the overall size and size diameter tolerance, should all help to produce the maximum possible accuracy.

Master gears should check production gears over the entire active portion of their involutes.

A master gear's tooth thickness should be known. The arc tooth thickness at the nominal pitch diameter is used for accurate center distance calculations.

Master gears should be heat treated to minimize subsequent distortions. Nevertheless, they should be checked from time to time for the effects of wear and abuse. Remember, inaccuracies in a master gear are costly.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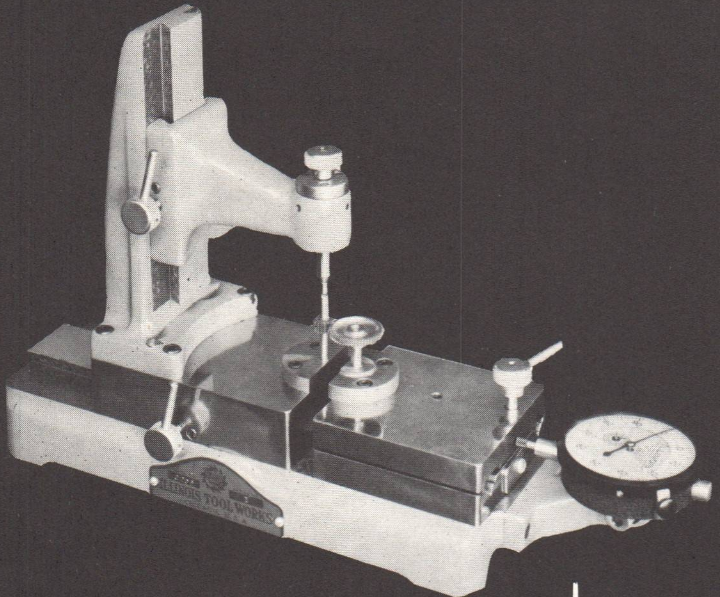

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